Condenser Optimization in Steam Power Plant

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In this paper the effects of the condenser design parameters (such as turbine inlet condition, turbine power and condenser pressure) on heat transfer area, cooling water flow-rate, condenser cost and specific energy generation cost are studied for surface type condenser. The results are given in the text and also shown as diagrams.

Keywords: condensers, steam power plants, energy cost.

Introduction

The main purpose in steam power plant is to generate maximum power at high efficiency. The most effective way for reaching this aim is to use a condenser. Condensers increase the enthalpy drops and turbine work by lowering the turbine outlet pressure. The lower the pressure, the higher the efficiency and power are. Condensers can be classified in two groups: surface and direct contact. At present, while the first type is mostly used in power plants, the other is only used in special cases. In this paper we investigated the surface type condenser.

The principal cycle of the steam engines is Rankine Cycle. A schematic of a simple steam power plant with condenser and its h-s diagram are shown in Fig.1. Rankine Cycle is an ideal cycle where all the process take place as reversible. Actual cycle differs from ideal cycle because of some irreversibilities. Thermal efficiency of the actual cycle according to Fig.1 is:

$$\eta_{th} = [(h_1 - h_2) - (h_4 - h_3)] / (h_1 - h_4) \qquad (1)$$

and the turbine power:

$$P_e = m_s (h_1 - h_2)\, \eta_m \quad \text{(kW)} \qquad (2)$$

where h is the enthalpy in (kJ/kg), m_s the steam mass flow rate in (kg/s) and η_m the mechanical efficiency of the turbine.

As seen in equations 1 and 2, to maximize the thermal efficiency and power, the enthalpy drop in the turbine must be increased. There are mainly two ways to reach this aim. First one is to increase the turbine inlet temperature and pressure, the other one is lowering the turbine exit pressure, in other words to use a condenser. If the exit pressure is dropped from 0.1×10^5 Pa to 0.05×10^5 Pa the efficiency and power rise 7% approximately when the turbine inlet conditions remain constant.

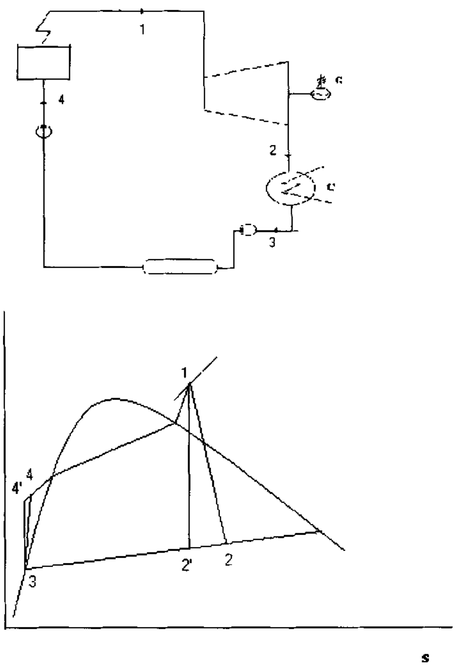

Fig.1 A schematic of a simple steam power plant and it is h-s diagram

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Condenser Calculations

From Fig.1 and Fig.2, the heat from the exit steam is:

$$Q_s = m_s (h_2 - h_3) \quad \text{(kJ/kg)} \tag{3}$$

and the heat that cooling water gets:

$$Q_w = m_w c_{pw} (T_a - T_i) \quad \text{(kJ/kg)} \tag{4}$$

From the equalization of Q_s and Q_w we get:

$$m_w = [m_s (h_2 - h_3)] / [c_{pw} (T_a - T_i)] \tag{5}$$

where m_w denotes the cooling water flow rate in (kg/s), c_{pw} the specific heat of the water in (kJ/kgK) and (T_a-T_i) the temperature difference of the cooling water. (T_a-T_i) can be taken as $(6 \div 8)$°C.

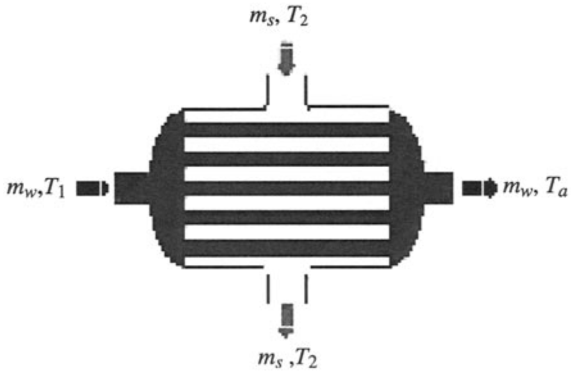

Fig.2 A schematic of a surface type condenser

The heat transferring in condenser is:

$$Q = A_1 \cdot k \cdot \Delta T_m \quad \text{(kW)} \tag{6}$$

where k is the heat transfer coefficient in (W/m^{2}K), A_1 the heat transfer area of the condenser in (m^2) and ΔT_m the log mean temperature difference. From Fig.3:

$$\Delta T_m = [(\Delta T_i - \Delta T_e)] / [\ln(\Delta T_i / \Delta T_e)] \tag{7}$$

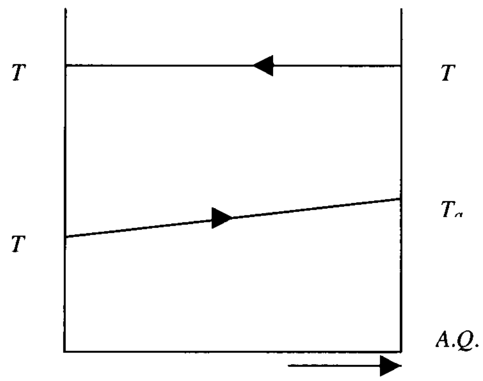

Fig.3 Heat transfer diagram of a condenser

ΔT_i is difference between saturation steam temperature and cooling water inlet temperature ($\Delta T_i = T_2 - T_i$), ΔT_e is difference between saturation steam temperature and cooling water exit temperature $\Delta T_e = (T_2 - T_a)$.

From Eq.6 condenser heat transfer area is:

$$A_1 = Q / (k \cdot \Delta T_m) \quad (\text{m}^2) \tag{8}$$

On the other hand some air enters condenser through different ways. It decreases the k value, increases the condenser pressure and causes the drops in efficiency and power. For this reason the air is removed by using a vacuum pump or ejector. This air also cooled by cooling water so that it needs additional condenser area. This area is added into A_1. The cooling area of air can be computed from the equation given below.

$$A_2 = (m_a c_{pa} / k_1)\{\ln[(T_2 - T_i)/(T_u - T_i)]\} \quad (\text{m}^2) \tag{9}$$

where m_a refers to air mass entering the condenser per hour, k the heat transfer coefficient from air to water and T_u the air inlet temperature.

According to Hoeffer[1], m_a can be taken as:

$$m_a = 0.0314\,(m_s/10^2)^{0.9} \quad \text{(kg/s)} \tag{10}$$

As a result the total area is $A = A_1 + A_2$.

For a safety cooling, this value of A is increased from 10 percent to 25 percent, so that the final area A_t is between 1.10 A and 1.25 A.

If we take the length of the cooling water tubes as L, inner diameter of the tube as d_i, outer diameter as d_a and tube number as n, A_t can be written as:

$$A_t = \pi \cdot d_a \cdot L \cdot n \quad (\text{m}^2) \tag{11}$$

Cooling water velocity in the tubes:

$$v = m_w / [(\rho \cdot d_i^2 \cdot n)(\pi/4)] \quad \text{(m/s)} \tag{12}$$

where ρ is the density of water in (kg/m^3).

Cooling water heat transfer coefficient is:

$$\alpha_w = (a \cdot v_w^n / d_i^m)(1 + b \cdot T_m) \quad (\text{W/m}^2\text{K}) \tag{13}$$

where T_m is the mean temperature of the cooling water and a, b, m, n are the constant factors. From Hutte, we can take a=1755, b=0.015, n=0.87 and m=0.13. If we denote the dirtiness factor as φ then[2]:

$$\alpha_w^* = \varphi \cdot \alpha_w \quad (\text{W/m}^2\text{K}) \tag{14}$$

φ value can be taken between 0.60 and 0.70.

The overall heat transfer coefficient k can be drawn from equation below[3]:

$$1/k = (1/\alpha_w^*) + (\delta/\lambda) + (1/\beta_s) \tag{15}$$

where δ is the tube thickness in m, λ the thermal conductivity of the tube material in (W/m^{2}K) and α_s the heat transfer coefficient of the steam (W/ m^{2}K) α_s value is taken between 11000~16000 (W/m^{2}K).

In condenser calculation, the heat transfer coefficient k is assumed firstly, calculation are done according to this assumption and finally k is controlled. k value varies from 2000 (W/m^{2}K) to 3500 (W/m^{2}K) in turbine condensers.

The power of the condenser cooling water pump is:

$$P_p = (0.1 \cdot m_w \cdot \Delta P) / \eta_p \quad \text{(kW)} \tag{16}$$

where η_p is the pump efficiency and ΔP the total pressure loss.

Condenser Cost

Condenser cost equation can be written as below:

$$C_{\text{con}} = A_t \cdot g_c \cdot \beta \quad (\$) \tag{17}$$

where C_{con} is the condenser cost, A_t the total condenser heat transfer area (m^2), g_c the specific condenser cost (\$/kg) and β the relation between condenser heat transfer area and condenser mass. The share of condenser cost in energy generation cost,

$$g_{\text{con}} = C_{\text{con}} \cdot / E \quad (\$/\text{kWh}) \tag{18}$$

Energy Generation Cost

Electrical energy generation cost consists of three items[4]:

$$C_t = C_k + C_f + C_m \quad (\$) \tag{19}$$

where C_t is the total cost, C_k the capital cost, C_f the fuel cost and C_m the operation and maintenance cost per annum. If we divide the total cost by annual electrical energy generation we obtain specific energy cost:

$$g_e = C_t / E = (C_k + C_f + C_m)/E \quad (\$/\text{kWh}) \tag{20}$$

Annual electrical energy generation can be written as:

$$E = 8760 \cdot P_e \cdot n_k \quad \text{(kWh)} \tag{21}$$

where P_e is the plant power and n_k is the capacity factor.

Conclusion

In this paper, the effect of condenser pressure and plant power on condenser cost and specific energy generation cost are studied. For this aim, we used a computer programme running on Matlab R12 base. Results are given as diagram in Figs.4~6. Fig.4 shows the effect of the condenser pressure on heat transfer area. As seen from that, heat transfer area decreases with condenser pressure. In Fig.5 the effect of the cooling water velocity on heat transfer coefficient for different tube diameter is shown. As can be seen from Fig.5 heat transfer coefficient increases with the cooling water velocity. The effect of the condenser pressure on unit

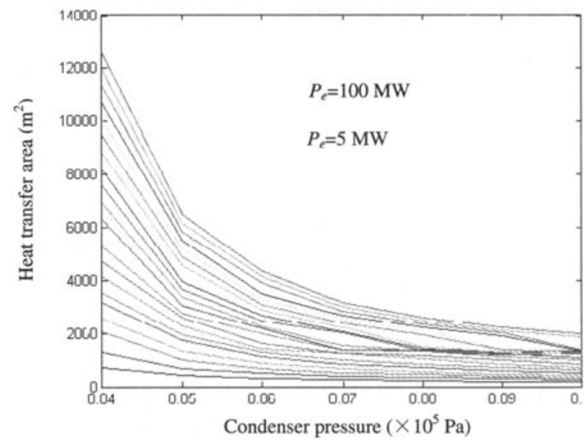

Fig.4 The effect of the condenser pressure on the heat transfer area

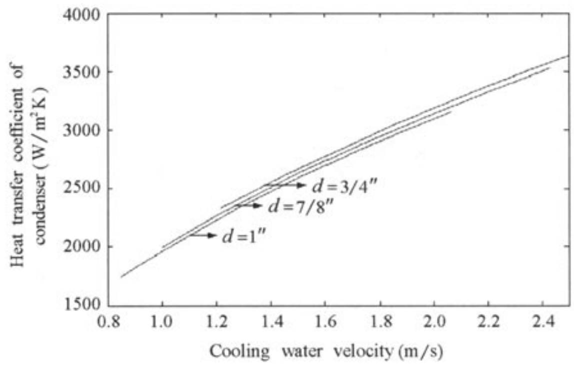

Fig.5 The effect of the cooling water velocity on heat transfer coefficient for different tube

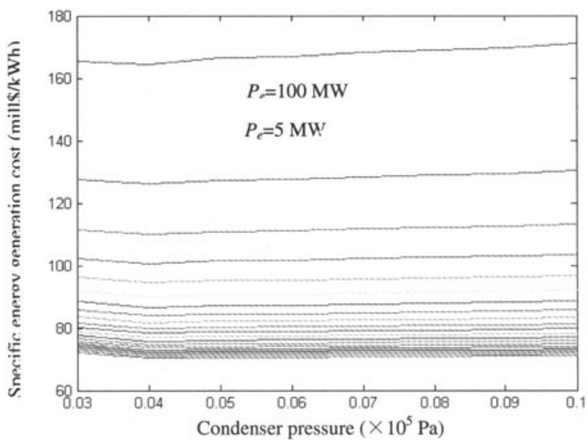

Fig.6 The effect of the condenser pressure on the unit energy cost

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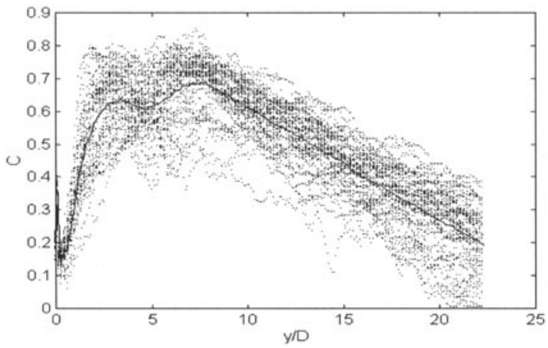

Fig.5 Concentration of the reaction product in terms of vertical distance from the outlet of the nozzle

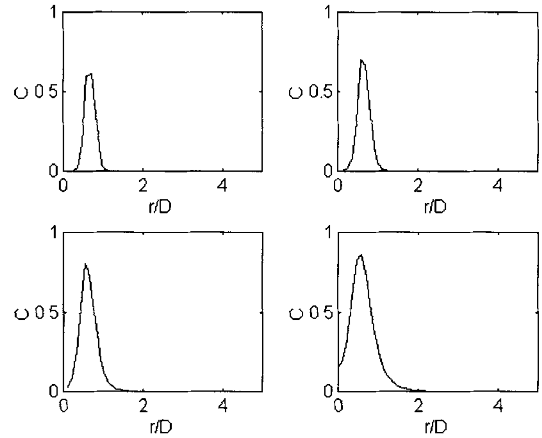

Fig.6 Numerical results of the mean relative concentration of NH_4Cl aerosol versus distance from the symmetry axis in different cross sections
The distance from the outlet of HCl:
(a) 1D (b) 2D (c) 5D (d) 10D

The mean concentration function has two local maximum in approximately 3D and 7D distance from the outlet.

Numerical Approach

The numerical modelling of the reaction was performed by means of Fluent code. The finite rate chemistry approach was used together with LES model for turbulence. The calculation were performed for boundary conditions from the experiment. The space size of the numerical grid node in the vicinity of the nozzle outlet was 0.05 mm. The profiles of mean relevant concentration of the reaction product is exhibited in Fig.6. The numerical calculations can give only averaged results without fluctuations, because the physical model is based on the averaged balance equations.

In the light of the experimental observation the rate constant k proposed by Countess and Heicklen[2] seems to be too low, because the numerical concentration in terms of vertical space coordinate is monotonically growing up instead of reaching maximum in a distance of 5~8D from the outlet.

Acknowledgements

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energy cost is given for different power in Fig.6. ccording to that, specific energy cost decreases with condenser pressure, but changing is fairly slight. Fig.4 and Fig.6 show that especially below 0.05×10^5 Pa condenser pressure, while the heat transfer area rises sharply, specific energy generation cost increases slightly. On the other hand, since the analytical method is used with this study, if one changes the parameter another solution may be obtained.

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